EFFECT OF RETURN AIR LEAKAGE ON AIR CONDITIONER PERFORMANCE IN HOT/HUMID CLIMATES

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ABSTRACT
An experimental study was conducted to quantify the effect of return air leakage from hot/humid attic spaces on the performance of a residential air conditioner. Tests were conducted in psychrometric facilities where temperatures and humidities could be controlled closely.

Return air leakage from hot attic spaces was simulated by assuming adiabatic mixing of the indoor air at normal conditions with the attic air at high temperatures. Effective capacity and Energy Efficiency Ratio both decreased with increased return air leakage. However, power consumption was relatively constant for all variables except outdoor temperature, which meant that for the same power consumption, the unit delivered much lower performance when there was return air leakage. The increase in sensible heat ratio (SHR) with increasing leakage showed one of the most detrimental effects of return air leakage on performance.

INTRODUCTION
In recent years, many electric utilities have provided rebates to residential customers for purchasing high efficiency air conditioners and heat pumps. The rebates have helped increase the demand for higher efficiency air conditioning units. However, even the most efficient system will not perform as expected if it is not installed properly. Installation and maintenance items such as improper amount of charge in the system, reduced evaporator airflow, and air leakage in the return air duct from a hot attic space are very important in the determination of the performance of these units at high outdoor temperatures.

The purpose of this paper was to experimentally quantify the effect of air leakage in the return air duct from a hot attic space on the high-temperature performance of air conditioners and heat pump systems. Air conditioner performance is quantified in terms of capacity, Energy Efficiency Ratio (EER), power, and sensible heat ratio (SHR).

PREVIOUS WORK
Leakage in residential air distribution systems may have a large impact on energy consumption, peak utility demands, and ventilation in a significant fraction of houses in the United States (Modera, 1989). In regions of the U.S. in which duct systems pass through unconditioned spaces, air infiltration rates into a residence will typically double when the distribution fan is turned on. Also, average annual air infiltration rate is increased by 30% to 70% due to the existence of the distribution system (Modera, 1989).

According to Modera (1989), increases in peak electricity demands due to duct leakage can be as high as four kW per house in hot and humid climates, assuming that the return ducts pass through the attic. Increases in peak load per house on the order of one to two kW are likely in less extreme climates, or with less extreme return duct conditions. Using a simplified analysis, duct leakages in Sacramento, CA, and West Palm Beach, FL, were calculated to cause between two and ten MWh/yr increases in annual energy consumption. These results should be applicable to most of the Southern United States. It was observed that return-side leakage represented a surprisingly large fraction (64%) of total duct leakage. Their results showed that annual energy consumption was strongly dependent on the location of the return duct. Also, it was observed that a system with an attic return consumed between one to five MWh/yr more energy than a system with a crawl-space return.

Lambert and Robison (1989) tested four different groups of houses for air leakage. There were two control groups and two test groups. The control groups were houses built according to current regional practice. The test groups consisted of homes built under the highly energy-efficient Model Conservation Standard (MCS). This standard required substantially above-code insulation, infiltration reduction measures, heat recovery ventilation, and well insulated ducting. Both the control and test groups were then subdivided in two more groups, houses with or without forced-air heating systems. Their results showed that homes with ducted forced-air heating had more whole-house air leakage than homes without ducting. That air leakage difference averaged 26% for current practice homes. The incremental leakage and thermal losses due to the presence of ducting were substantially lower in highly energy-efficient homes (MCS). Incremental leakage and thermal losses for the MCS ducted group were 22% and 13%, respectively.

Their results suggested that current construction practices associated with forced-air heating systems should be reviewed (Lambert and Robison, 1989). Substantially better performance was obtained by the ducted MCS test group. However, even the MCS test group had much greater leakage than the control homes (unducted). The authors agreed that leakage of the ductwork itself was a significant part of the problem, and that conduction losses from ducting must have played some role in the thermal losses.

In a companion study of more than 20 ducted homes (Robison and Lambert, 1989), measured leakage of return ducts was found to be about twice the amount of supply ducts (Table 1). The predominance of return duct leakage over supply duct leakage confirmed the investigators' theory that installers were more careful to seal seams on supply ducts (Robison and Lambert, 1989).

Table 1 - Measured Leakage and Percent Increase in Total House Leakage Due to Return and Supply Ducts

Duct	Measured Leakage (cfm)	% Increase in Total House Leakage
Supply	86.2	3.9
Return	178.1	6.4

* (Source: Robison and Lambert, [1989])

According to Cummings and Tooley (1989), return leakage of 15% attic air can reduce air-conditioner capacity by 50% or more during the peak cooling hours of the day, when maximum capacity is needed. Twenty percent return leaks of outdoor air can increase cooling season energy use by about 20%. If the source of the leak is attic air, the increase may be as much as 100%. They agreed with Robison and Lambert's research in that leaks in the return air ducts are typically larger than supply duct leaks. The importance of sealing supply leaks is obvious, because conditioned air is being lost. However, the problems of return leaks are not widely recognized. Energy savings from duct repair can be quite large (Cummins and Tooley, 1989). A 20% return leak coming from outdoor air at 95°F (35°C) dry-bulb temperature and 75°F (23.9°C) dew point temperature would reduce the net air conditioner capacity by about 30%. If this same return leak comes from an attic at 130°F (54.4°C) dry-bulb temperature and 85°F (29.4°C) dew point temperature, then the net air conditioner capacity may be reduced by 95%.

The cost of repairing these systems is quite low. Based on estimates from a Florida company which specializes in these repairs, the cost for diagnosing the house and duct system should be about $50. The cost for the repair itself is typically less than $100 ($60 for a simple fix and as much as $150 for a more complex repair). More research is needed to evaluate how widespread the problem of air leakage is, to determine ways of preventing and correcting return leaks, and to assess the energy and peak demand penalties created by these leaks.

EXPERIMENTAL APPARATUS

The objective of the experiments was to study the effect of evaporator airflow, refrigerant charge, and return air leakage on the performance of residential air conditioners and heat pumps. The data collected included refrigerant and air mass flow rates, pressures and temperatures of refrigerant throughout the system, and power consumption of the unit. The experimental apparatus consisted of (1) the psychrometric rooms, (2) indoor and outdoor test sections, (3) the test air conditioners, and (4) installation and data acquisition.

The air conditioning units were tested in the psychrometric rooms of the Energy Systems Laboratory at Texas A&M University. These rooms can control temperature and humidity for both indoor and outdoor sections of the unit. Dry-bulb and wet-bulb temperatures can be maintained within ±0.2°F during steady-state operation. The rooms were designed for testing systems with cooling capacities of up to 10 tons.

The unit used for return air leakage tests was a 3.5 ton (12.3 kW) split system air conditioner with TXV expansion and a scroll compressor. It had a seasonal energy efficiency ratio (SEER) of 13. Conditions at the entrance to the evaporator were varied to simulate the hotter and more humid conditions that could be expected when there is air leakage from a hot attic into the return air duct. The effect of return air leakage on capacity, power, and energy efficiency ratio (EER) of an air conditioner were quantified. The air conditioner was subjected to an outdoor temperature of 100°F (37.8°C). The indoor conditions for the no leakage test were set at 75°F (23.9°C) dry-bulb temperature and 50% relative humidity (RH).

The simulated leakage amounts depended on the expected conditions in the attic space. With assumed attic space temperatures of 130°F and 150°F, the amount of leakage (on a mass basis) was varied between 4.7% to 20%. For these same attic temperatures, the attic relative humidities were varied between 10% and 35%. Attic temperatures between 130 and 150°F are not uncommon in houses in the summer. Attic humidities will vary depending on the location of the house. In southern coastal cities, the attic humidities can be much higher than in the desert southwest. This research focused on high attic humidities.

The indoor test section consisted of the indoor air flow chamber, the test section, and the indoor coil. Conditioned air from the indoor psychometric room passed through the indoor fan coil unit, insulated airflow ductwork, and the indoor airflow chamber. A 12-element thermocouple grid placed at the entrance of the fan coil unit measured entering air dry bulb temperature. Another 12-element thermocouple grid was used at the exit of the unit to provide exit dry-bulb and wet-bulb conditions. Figure 1 shows a sketch of the test section and indoor coil.

A booster fan located at the exit of the flow chamber was used to draw air through the air duct. Air velocity through the test section was controlled by dampers located on the exit of the booster fan. The outdoor test section consisted of the outdoor fan coil unit. Air was drawn in through three sides of the outdoor unit and discharged through the top of the unit. A sampler mounted around the outdoor coil measured dry-bulb and dew-point temperatures.

The instrumentation for all tests was divided into air-side, and refrigerant-side measurements. The air-side temperature measurements for both the inlet and outlet of the indoor coil unit were made using 12-element type-T thermocouple grids. Wet-bulb sensors were also used for both the inlet and outlet of indoor coil unit. For the outdoor unit, the only air-side temperature measured was the inlet air temperature. It was measured with a single type-T thermocouple located in the sampling duct surrounding the outdoor unit.

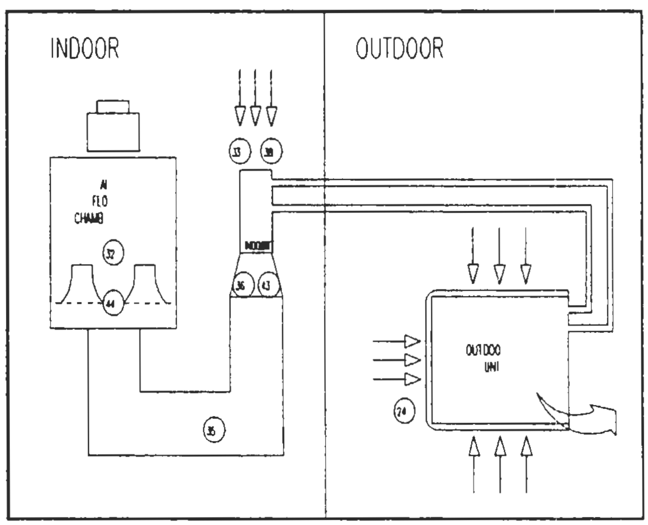

Figure 1 - Layout of the Psychrometric Rooms Showing the Placement of the Indoor and Outdoor Units (Numbers indicate measurement points.)

The refrigerant-side measurements consisted of temperature and pressure measurements throughout the refrigerant lines. The refrigerant-side test points are shown on Figure 2. With the exception of the refrigerant flow rate, the outdoor unit power, and the air flow differential pressure, the rest of the measurements were temperatures (dry-bulb, wet-bulb, dew point) and pressures.

At each point that a temperature measurement was taken, pressure transducers were used to measure refrigerant pressure. Refrigerant mass flow was measured with a Coriolis-type mass flow meter. The flow meter was placed on the liquid line after the condensing unit (Figure 2).

The data acquisition system converted signals coming from all the sensors in the indoor and outdoor rooms into temperatures, pressures, flow rates, or power. A data logger was used to collect data from the testing apparatus. The logger was linked to a computer where the data were visually displayed during testing. Once a test was complete, the data were transferred to another computer for processing. A total of 22 channels were monitored during testing. Each channel was scanned by the logger at 30 second intervals.

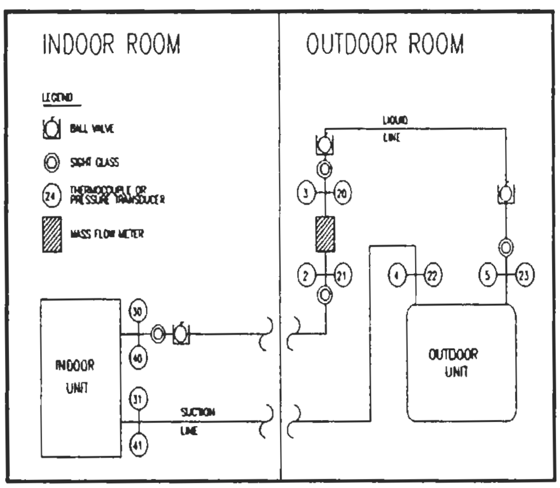

Figure 2 - Data Acquisition Points in the Refrigerant Side of the System (Numbers indicate measurement points.)

The parameters used to describe the performance of the air conditioners were capacity, EER, power consumption, and SHR. To calculate capacity, it was necessary to know the enthalpies leaving and entering the evaporator, as well as the air flow rate. The evaporator inlet and exit temperatures and pressures were used to determine the values for enthalpy. Refrigerant temperatures and pressures at the evaporator and expansion device were used to determine the values for superheat and subcooling respectively. The EER was calculated from the capacity and power measurements.

TEST PROCEDURE

Figure 3 shows the leakage in a return air duct. Air at state 1 is pulled into the duct system from the conditioned space. With improperly sealed ducts, air leaks in at state 2 and mixes with the normal return air to provide conditions at state 3 entering the fan coil. The conditions at state 3 depend on the quantity and state of air at state 2. The capacity of the air conditioner at the coil could actually increase with the addition of the leakage air from state 2. This leakage air would increase the entering air temperature and humidity at state 3, which would increase capacity. However, the effective cooling effect on the residence would decrease because the exiting temperature at state 4 would increase over what it would have been without leakage because of the hot air leaking into the return air ducts. Thus, the introduction of return air leakage adds a complexity to the calculation of capacity and requires defining some new terms. First, the effective capacity can be defined as:

$$CAP_{eff} = \dot{m}_{Air1}\, h_1 - \dot{m}_{Air4}\, h_4 - P_{fan} \qquad (1)$$

where:

$\dot{m}_{Air1}$ = mass flow rate of air entering the evaporator (Baseline Tests)

$\dot{m}_{Air4}$ = mass flow rate of air exiting the evaporator (Modified Tests)

h_1 = enthalpy of air entering the evaporator (Baseline Tests)

h_4 = enthalpy of air exiting the evaporator (Modified Tests)

P_{fan} = heat addition caused by the fan

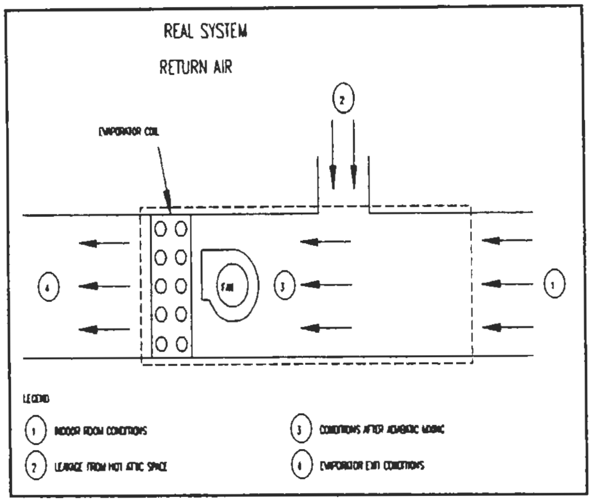

Figure 3 - Evaporator with Return Air Leakage from Hot Attic Space

This unit did not have a fan. According to the ARI test procedures (ARI, 1989), for the capacity calculations, a theoretical value for heat addition (365 W/1000 cfm) must be taken into account. Thus, the effective capacity calculated in Equation 1 must be decreased by the amount of heat generated by the fan in the airstream. After the effective capacity was calculated, the effective energy efficiency ratio (EER_{eff}) was calculated by dividing the effective capacity by the total power consumption values from the modified tests.

$$EER_{eff} = \frac{CAP_{eff}}{P_{tot}} \qquad (2)$$

Note that the amount of air entering from the conditioned space, m_{air1}, can be different from the amount of air introduced into the conditioned space, m_{air4}. Under conditions with no leakage, these two mass flows would be the same and the effective cooling capacity to the residence would be equal to the measured cooling capacity of the unit. However, with leakage from an attic space, there is introduction of higher temperature air and moisture. If the leakage rate is high enough or attic conditions are moist enough, the outlet humidity at state 4 may be higher than state 1. Under these conditions, the air conditioner would be providing no effective latent cooling to the residence.

One difficulty arises in the laboratory on how to simulate the leakage into a real return duct. Figure 4 shows the approach used here. In normal tests (top case in Figure 4), air at the given indoor room conditions (75°F (23.9°C) and 50% relative humidity) was drawn through the fan coil unit and data taken. Rather than attempting to create conditions corresponding to state 2 and mixing these with state 1, a modified test (bottom case in Figure 4) was run. The conditions that would exist at state 3 were first calculated. For example, mixing 87% air at of 75°F (23.9°C) and 50% RH, with 13% attic air at 130°F (54.4°C) and 10% RH would produce a mixed air condition at state 3 of 82.6°F (28.1°C) DB, and 65.0°F (18.3°F) WB temperatures. For these tests, the indoor environmental room was set to state 3 conditions and tests completed. The outlet conditions for these tests provide the state 4 conditions in the above equation for effective capacity. State 1 conditions were provided by the normal return air conditions of 75°F (23.9°C) and 50% relative humidity.

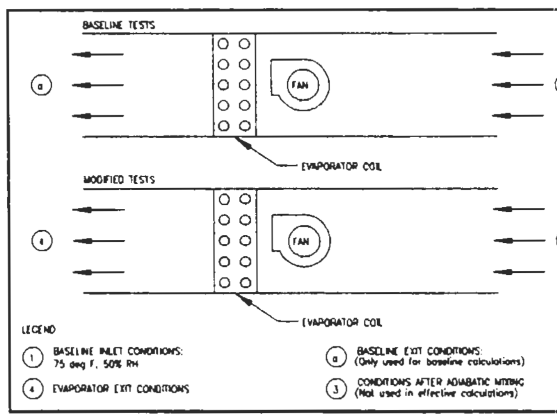

Figure 4 - Baseline and Modified Tests Used to Simulate the Effects of Return Air Leakage

Because conditions at state 3 in Figures 3 and 4 represented an adiabatic mixing process, there were an infinite number of temperature and humidity conditions at state 2 that could be used with the incoming air at state 1 to produce the mixed temperatures. Thus, from a relatively small number of tests with the mixed conditions at state 3, it would be possible to predict the impact of leakage over a wide range of conditions in the attic space. Take for example, the condition discussed in the previous paragraph. The mixed conditions at state 3 of 82.6°F (28.1°C) DB, and 65.0°F (18.3°C) WB temperatures were assumed to be produced by mixing 87% air at 75°F (23.9°C) and 50% RH, with 13% attic air at 130°F (54.4°C) and 10% RH. Table

2 shows other attic conditions that would produce the same conditions at state 3. If the air temperature were 150°F and 6.1% humidity, the amount of attic leakage would only be 9.6% rather than the 13% required by the 130°F and 10% relative humidity air. Other conditions could be calculated but attic temperatures from 110°F to 150°F on a hot summer day should easily represent values found in most parts of the country.

Table 2 - Attic conditions and leakage needed to produce the same mixed air temperature entering the evaporator.

Attic Conditions		% Attic Leakage	% Return Air
Temp (°F)	Humidity (%)		
110	17.2	20.4	79.6
130	10	13.0	87.0
140	7.7	11.0	89.0
150	6.1	9.6	90.4

RESULTS

The experimental results were expressed in terms of effective capacity, power and energy efficiency ratio, and sensible heat ratio. Each is discussed below.

<u>Effective Capacity</u>

The capacity dropped as the amount of air leakage into the return air duct and attic humidities increased. The maximum capacity value of 37,663 Btu/h (11.0 kW) occurred at 0% leakage (Figure 5). For example, at 100°F (37.8°C) outdoor temperature and 130°F (54.4°C) attic temperature, the capacity dropped 19.2% for 10% relative humidity as the leakage from the attic increased from 0% to 13%. Figure 6 shows that the drop was larger at 150°F (65.6°C). For example, at 100°F (37.8°C) outdoor temperature and 150°F (65.6°C) attic temperature, the capacity dropped 25.8% for 10% relative humidity as the leakage increased from 0% to 9.1%. For the same leakage, an increase in humidity to 14.2% yielded a drop in capacity of 39.5%. Attic conditions, especially attic humidity, were important factors on the decrease in capacity with increased air leakage.

As discussed earlier, it was possible to take a single test with a fixed mixed air temperature and humidity and use it to estimate the impact of a variety of leakage air temperatures, humidities and flow rates. Figure 7 shows a plot of leakage attic air temperatures ranging from 110°F to 150°F. The high attic humidity conditions for this plot have dew point temperatures ranging from approximately 70°F to 82°F. As seen in these three figures, increasing the amount of moisture in the airstream had a significant impact on the effective capacity of the air conditioner.

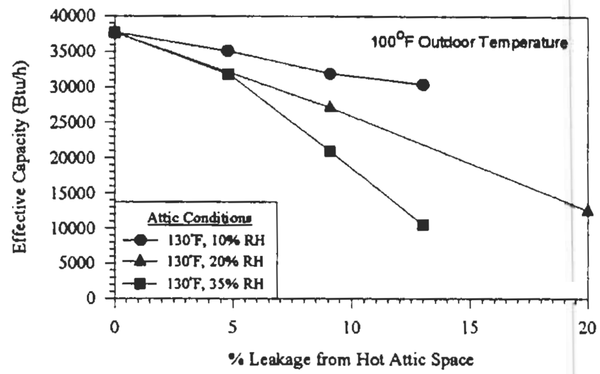

Figure 5 - Effective Capacity at 130°F (54.4°C) Attic Temperature and 100°F (37.8°C) Outdoor Temperature for Various Attic Humidities and Leakage Conditions.

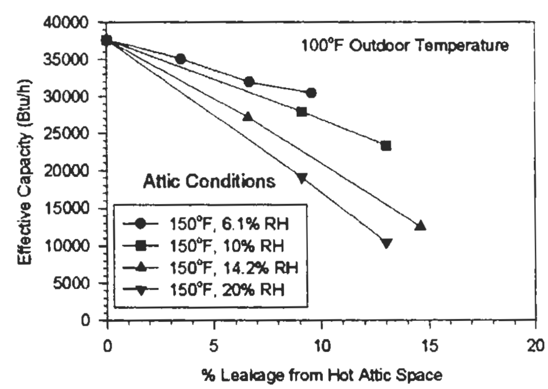

Figure 6 - Effective Capacity at 150°F (65.6°C) Attic Temperature and 100°F (37.8°C) Outdoor Temperature for Various Attic Humidities and Leakage Conditions.

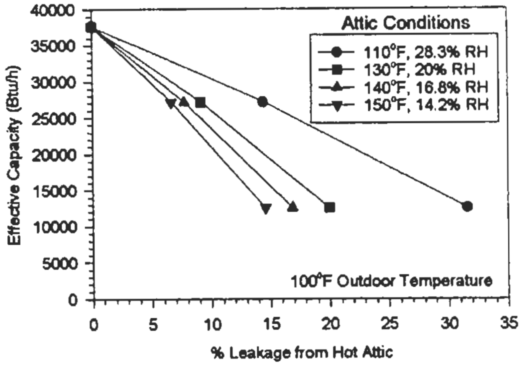

Figure 7 - Effective Capacity at Different Attic Temperature and a Range of High Attic Humidity Conditions

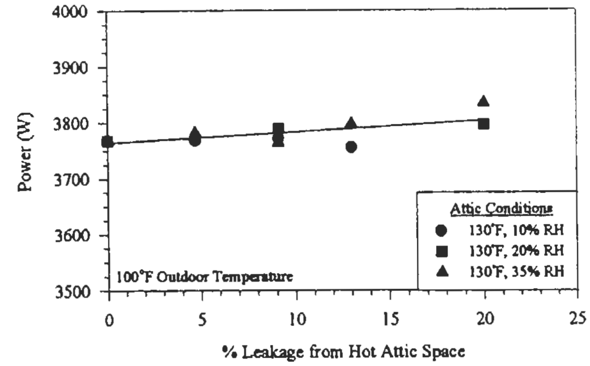

Figure 8 - Power Consumption at 130°F (54.4°C) Attic Temperature and 100°F (37.8°C) Outdoor Temperature for Various Attic Humidities and Leakage Conditions

Power Consumption

The power consumption was relatively constant for all conditions except for increasing outdoor temperatures, where it increased (Figure 8). Power consumption for 100°F (37.8°C) outdoor temperature increased by less than 70 watts over the baseline of 3.77 kW as the leakage rate increased from 0 to 35%. At 120°F (48.9°C), the power was always within 1.5% of 4.56 kW. This small increase in power would indicate that leakage should have little impact on the peak demand from a residence.

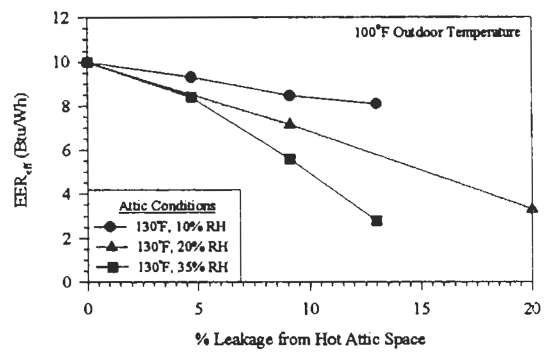

Figure 9 - Effective EER at 130°F (54.4°C) Attic Temperature and 100°F (37.8°C) Outdoor Temperature for Various Attic Humidities and Leakage Conditions

Effective Energy Efficiency Ratio

The EER_{eff} (Btu/Wh) was calculated by dividing the effective capacity across the evaporator (Btu/h) by the power input (W). The EER_{eff} showed similar trends to the capacity curves because the power consumption was relatively constant at a given outdoor temperature. The EER_{eff} dropped as the air leakage into the return air duct increased (Figure 9). The EER_{eff} also dropped for increased attic humidities. At 100°F (37.8°C) outdoor temperature and 130°F (54.4°C) attic temperature, the EER_{eff} dropped 15.3% for 10% relative humidity from 0% leakage to 9.1% leakage. It increased to 28.3% drop for 20% relative humidity at the same attic and outdoor temperatures(Figure 9).

Figure 10 shows a plot of the effective EER for leakage air temperatures ranging from 110°F to 150°F for high attic humidity conditions (dew points ranging from approximately 70°F to 82°F). The impact of a small amount of leakage is dramatic. For an attic temperature of 130°F, a 9.1% leakage rate produced a drop in effective EER from 10.0 to 7.2. For the owner of the residence, this would represent an increase in air conditioning operating costs of as much as 38% if the air conditioner could effectively meet the load in the residence. If it could not meet the thermal load in the house, then it could also mean that the temperature (and humidity) levels in the house would rise. This figure also shows that if the attic temperature is as high as 150°F, then only a 6.6% leakage rate is required to produced the same decrease in effective capacity as the 9.1% leakage rate with the 130°F air.

For consumers, these results mean that with duct leakage, they would have higher costs for cooling. The air conditioners would have to run longer to meet loads because of the reduced cooling capacity and would run at a much lower cooling efficiency than they would have without leakage in the attic. For electric utilities, these results mean that residential customers' air conditioners would cycle less frequently. While the power demand of an individual house at high outdoor temperatures would be similar to that without leakage, the average peak demand for a large number of houses might increase because of less on/off cycling of the units.

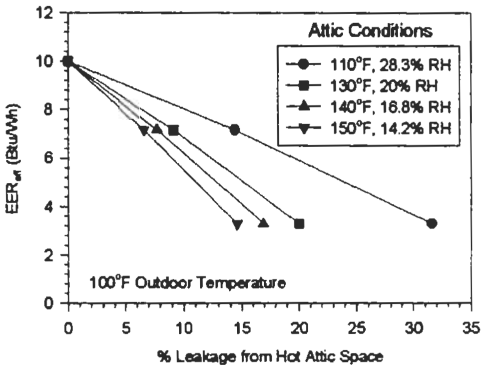

Figure 10
Effective Energy Efficiency Ratio at Different Attic Temperature and a Range of High Attic Humidity Conditions

Effective Sensible Heat Ratio

The effective sensible heat ratio (SHR_{eff}) was defined as the ratio of the effective sensible capacity to the effective total capacity. Under normal test conditions, the SHR would vary between 0 and 1. However, with the introduction of hot and potentially humid air into the return air duct, the air leaving the evaporator could have more water vapor in it than the air that is returned from the residence. Under these conditions, the SHR would exceed one (Figures 11 and 12). An SHR value greater than one would suggest a system that could provide the sensible cooling but not the dehumidification necessary for comfort.

Increased leakage from a hot attic caused an increase in sensible heat ratio. At 130°F and 10% attic relative humidity, however, the SHR_{eff} was relatively constant with increasing air leakage (Figure 11). For example, at 100°F (37.8°C) outdoor temperature and 130°F (54.4°C) attic temperature, the greatest difference in SHR_{eff} for 10% relative humidity from 0% leakage to 13% leakage was only 2.5%. In contrast, leakage at very high humidities (110°F and 45% relative humidity) produced an increase in SHR_{eff} from 0.8 at 0% leakage to 1.1 at 14.6% leakage. Because the SHR increased above 1.0, the unit would be unable to remove moisture from the air.

For fixed attic relative humidity, outdoor temperature, and amount of air leakage, increased attic temperatures caused an increase in effective sensible heat ratio (Figure 12). For 9.1% leakage, and at 130°F (54.4°C) attic temperature, the SHR_{eff} increased from 0.803 at 10% attic relative humidity to 0.93 at 20% attic relative humidity. In comparison, at 150°F (65.6°C) attic temperature, the SHR_{eff} increased from 0.879 at 10% relative humidity to 1.16 at 20% relative humidity.

For constant attic conditions of 150°F (65.6°C) temperature and 20% RH, increased outdoor temperatures caused a larger increase in SHR_{eff} with increasing return air leakage (Figure 9.20). At 100°F (37.8°C) outdoor temperature, the SHR_{eff} increased from 0.803 for no attic leakage to 1.764 for 13% leakage. In comparison, at 120°F (48.9°C) outdoor temperature, the SHR_{eff} increased from 0.827 for no leakage to 3.54 at 13% return air leakage.

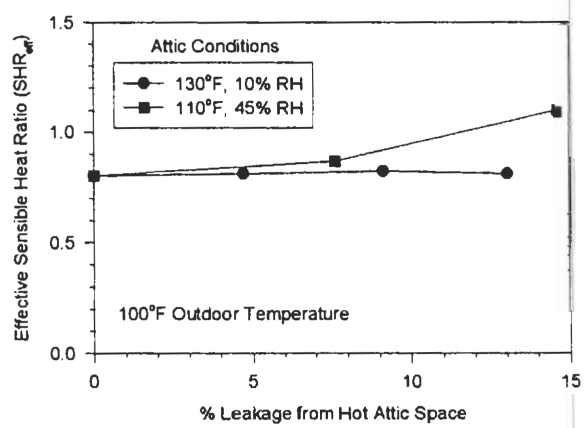

Figure 11 - Effective SHR at Two Attic Conditions and 100°F Outdoor Temperature for Various Attic Humidities and Leakage Conditions

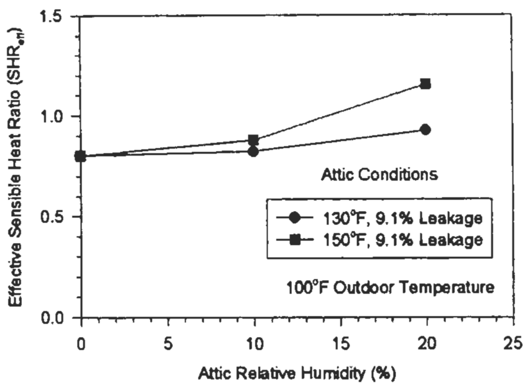

Figure 12 - Effective SHR at 10% Leakage and 100°F Outdoor Temperature for Various Attic Temperatures and Relative Humidities

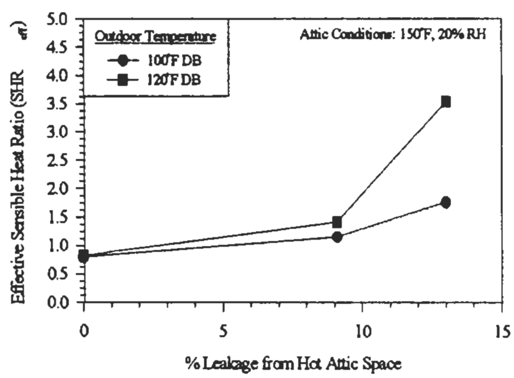

Figure 13 - Effective SHR at 150°F and 20% RH Attic Conditions for Various Outdoor Temperatures and Leakage Conditions

CONCLUSIONS AND RECOMMENDATIONS

This study focused on how leakage in the return air duct could affect the high temperature cooling performance of air conditioners and heat pumps. The primary performance variables examined included capacity, power, energy efficiency ratio, and the sensible heat ratio.

Effective capacity and EER_{eff} both decreased with increased return air leakage. The effect of leakage on capacity and EER_{eff} was larger for high humidity conditions, such as those found in coastal areas in the South than for the lower humidity conditions in the desert Southwest. Leakage amounts of as small as 5% in the high humidity areas can produce as high as a 20% reduction in both capacity and efficiency.

It is common in many houses to place the air conditioning evaporator in the hot attic space or in an unconditioned closet. If the return air ductwork is not sealed properly, there is an opportunity for conditioned air from the attic space or unconditioned closet to be pulled into the return air ducts. Results from the return air leakage studies suggest that even small amounts of air leakage can have a significant effect on the capacity, efficiency and dehumidification performance of the air conditioning system. For example, a 7.7% leakage rate from an attic space at 140°F and 16.8% relative humidity would cause a reduction in capacity and EER of approximately 28%.

Power consumption was relatively constant for all variables except outdoor temperature. While the unit would deliver less capacity to the residence, it would use the same power. The lower capacity would result in longer run-times of the air conditioners during warmer summer days. The longer run-times would reduce the diversity factor for electric utilities during warmer summer days. However, the total demand from an individual residence was independent of the amount of leakage within the range tested here. This conclusion appears to be in conflict with the result of at least one past study (Modera 1989).

The increase in SHR_{eff} with increasing leakage showed one of the most detrimental effects of return air leakage on performance. The leakage of enough hot/moist air can reduce the effective dehumidification the evaporator provides to the conditioned space. At high humidity conditions and with as low as 10% leakage, there were SHR values greater than one.

The overall effect of return air leakage on performance of the air conditioner system was more detrimental than expected. The effect of return air leakage is felt most during the hottest portions of the day when the need for air conditioning in the residence is highest. Return air leakage reduces the capacity of the air conditioner and requires the unit to run longer during the middle portion of a summer day. Unfortunately, for many electric utilities, the reduction in the performance of air conditioner happens to coincide to times of peak electrical demand.

For future work, there are several items that are suggested. First, a survey of residential air conditioners in different geographical areas could be made, focusing on checking for return duct leakage, measuring the amount of refrigerant charge in the units, and evaluating the amount of supply air through the evaporator. Each of these items may represent a larger benefit to consumers and electric utilities than a program that just promotes the installation of high efficiency air conditioners. While there have been a small number of studies on return attic leakage, these studies have typically covered a very limited sample. A broader study could provide information on how widespread these installation problems are and help the electric utilities determine if it is better to check for the proper installation of residential air conditioners rather than offer incentives to customers to buy new high efficiency units.

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